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THE STRUCTURE, GEOMETRY, AND KINEMATICS OF A UNIVERSAL JOINT

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ABSTRACT

The paper briefly presents the geometry, structure, and kinematics of a universal joint, very commonly used in machine building, especially today for heavy and engine-driven vehicles and transmissions located in different areas as well as for all-wheel-drive vehicles. The universal joint, or the cardan cross, conveys the rotation movement from one bridge to the other (when the rotary shaft suffers both movements, upward and downward). The kinematic scheme of a cardan transmission is composed of two cardan shafts (one input and one output), both of which are equipped with a cardan cross (universal joint or universal joint). Between the two universal couplings, a further (additional) cardan shaft (axle) is mounted. This mechanism is designed to transmit the mechanical movement (within a vehicle) from one bridge to the other. If the vehicle's motor is on the front and with on the rear axle transmission, or vice versa when the vehicle's engine is on the rear and the transmission is on the front axle, or when we have multiple (multi-axle) transmission on heavy vehicles or 4x4 cars.

Keywords: Cardan transmission; Universal joint; Heavy vehicles; Angular speed variation; Rotation movement; Geometry; Structure; kinematics.

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1. INTRODUCTION

The kinematic scheme of a cardan transmission can be seen in figure (1). It is composed of two cardan shafts (one input and one output), both of which are equipped with a cardan cross (universal joint or universal joint). Between the two universal couplings, a further (additional) cardan shaft (axle) is mounted. This mechanism is designed to transmit the mechanical movement (within a vehicle) from one bridge to the other. If the vehicle's motor is on the front and with on the rear axle transmission, or vice versa when the vehicle's engine is on the rear and the transmission is on the front axle, or when we have multiple (multi-axle) transmission on heavy vehicles or 4x4 cars (PETRESCU, 2012b).

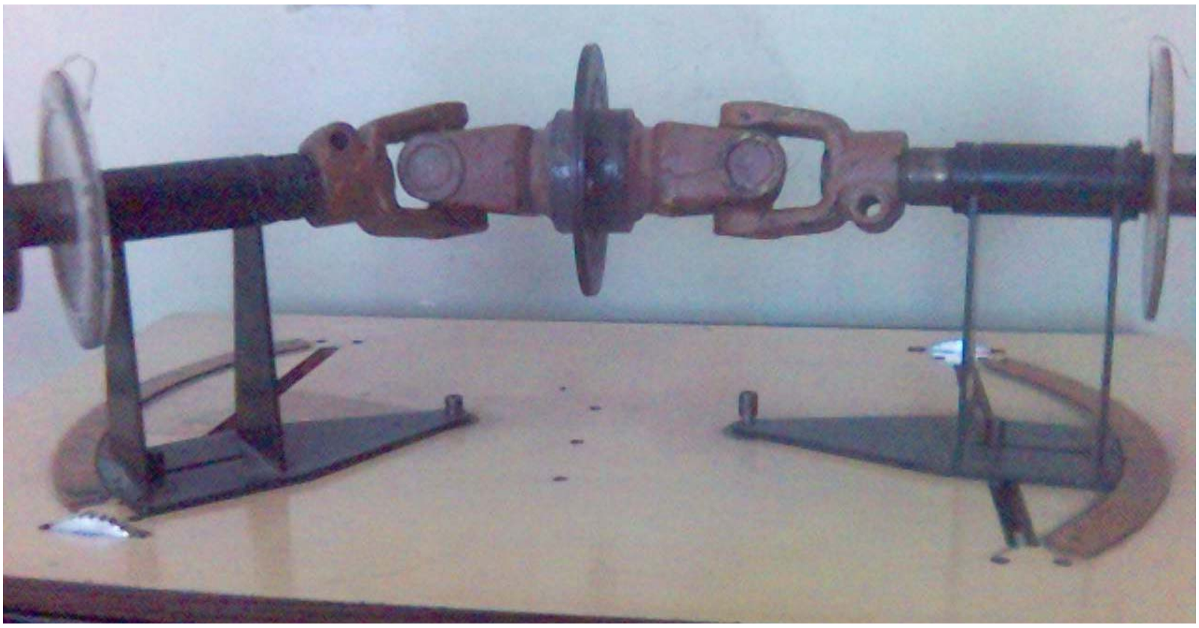

Figure 1: The cardan transmission

We most often encounter this transmission on buses, trucks, trolleybuses, trains, heavy cars, trucks, and all-wheel drive cars. It usually transmits the rotation movement from the front axle to the rear axle. It is necessary because there are big games in the transmission both left-right and up and down. It is the only mechanism that can transmit a rotation motion on a long axis that moves at the same time laterally and up and down. Generally, the yield of such a transmission is quite high, even if the rotation speed varies within the mechanism, but it is reconstituted at the end.

In 2010, more than 800 million vehicles circulate across the planet (ANTONESCU, 2000; ANTONESCU; PETRESCU, 1985; ANTONESCU; PETRESCU,

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1989; ANTONESCU et al., 1985a; ANTONESCU et al., 1985b; ANTONESCU et al., 1986; ANTONESCU et al.; ANTONESCU et al., 1987; ANTONESCU et al., 1988; ANTONESCU et al., 1994; ANTONESCU et al., 1997; ANTONESCU et al.; ANTONESCU et al., 2000a; ANTONESCU et al.2000b; ANTONESCU et al., 2001; AVERSA et al.; AVERSA et al., 2017a; AVERSA et al., 2017b; AVERSA et al., 2017c; ; AVERSA et al., 2017d; AVERSA et al., 2017e; MIRSAYAR et al., 2017; PETRESCU et al., 2017a; PETRESCU et al., 2017b; PETRESCU et al., 2017c; PETRESCU et al., 2017d; PETRESCU et al., 2017e; PETRESCU et al., 2017f; PETRESCU et al., 2017g; PETRESCU et al., 2017h; PETRESCU et al., 2017i, 2015; PETRESCU; PETRESCU, 2016; PETRESCU; PETRESCU, 2014; PETRESCU; PETRESCU, 2013a; PETRESCU; PETRESCU, 2013b; PETRESCU; PETRESCU, 2013c; PETRESCU; PETRESCU, 2013d; PETRESCU; PETRESCU, 2011; PETRESCU; PETRESCU, 2005a; PETRESCU; PETRESCU, 2005b; PETRESCU, 2015a; PETRESCU, 2015b; PETRESCU, 2012a; PETRESCU, 2012b; HAIN, 1971; GIORDANA et al., 1979; ANGELES; LOPEZ-CAJUN, 1988; TARAZA et al., 2001; WIEDERRICH; ROTH, 1974; FAWCETT; FAWCETT, 1974; JONES; REEVE, 1974; TESAR; MATTHEW, 1974; SAVA, 1970; KOSTER, 1974).

2. UNIVERSAL JOINT (CARDAN COUPLING)

The cardan coupling mechanism is a spherical mechanism (see figure 2) due to the spherical motion imposed by any universal coupler. In a cardan coupling the four rotation axes are competing in a point S (Figure 2); (PETRESCU, 2012b).

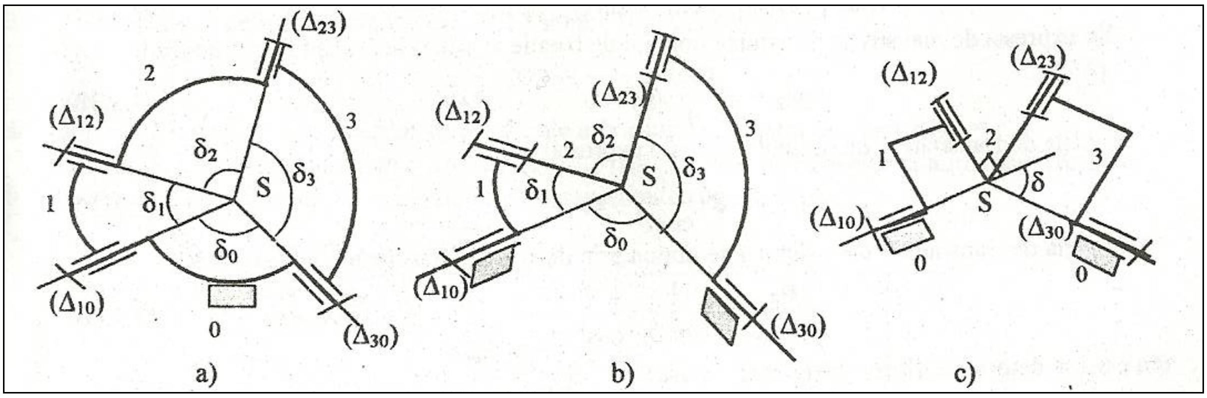

Figure 2: Universal joint or cardan coupling

The spherical mechanism is crank-type if the angle δ_1 is the smallest in relation to the angles δ_2, δ_3, δ_0 (figure 2a). In the case of the cardan mechanism, the element 2 (element representing a ball moving on a spherical surface with the center

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in S) is materialized by the two moving axes Δ_{12} and Δ_{23} (figure 2b). The specificity of the cardan mechanism is that the angles δ_1, δ_2, δ_3 are all 90^0 and the angle δ_0 between the two fixed axes is obtuse, the angle $\delta=\alpha$ (the attachment of δ_0) being sharpened (Figure 2c).

The analytical calculation of the input-output transmission function is done by means of the kinematic diagram of the cardanic mechanism (Figure 2c) and the schematic diagram of the u_1 and u_2 versions of the axes Δ_{12} şi Δ_{23}, where (figure 3a) is denoted by Δ_1 axis of the drive shaft and with the Δ_2 axis of the driven shaft. With these notations (Figure 3a), the transmission function between the input and output shaft (driven) is of the form $\varphi_2(\varphi_1)$ or $\psi_2(\varphi_1)$.

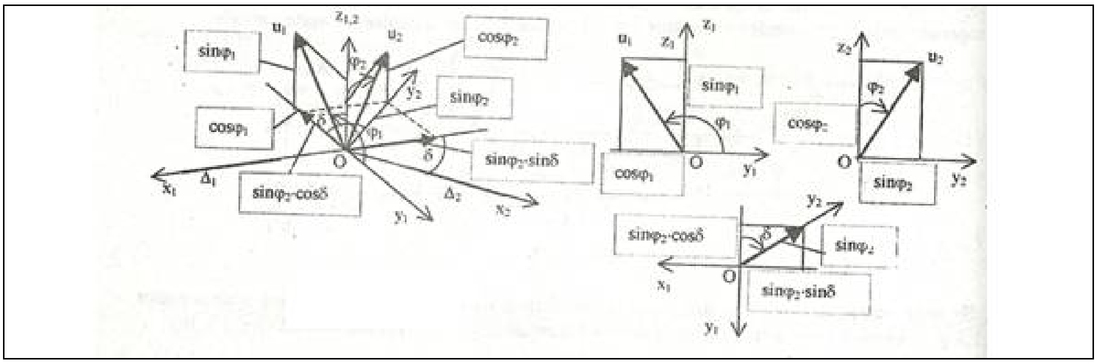

Figure 3: Cardan transmission: Versors of the transmit function

For this purpose one considers the mobile versor $\vec{u}_1$ and $\vec{u}_2$ (Fig. 3a), which are orthogonal and oriented in the directions of the moving axes Δ_{12} and Δ_{32} competing in the center O = S (Figure 2).

The versor $\vec{u}_1$ rotates in the plane [y_1z_1] and is positioned with the angle φ_1 from the y_1 axis (Figure 3). The vector $\vec{u}_2$ rotates with the angle $\varphi_2=\psi_2$ in the plane [y_2z_2], being positioned relative to the common axis $z_2 = z_1$ (Figure 3b).

The two versors are analyzed analytically by their components on axes x_1, y_1 and $z_{1,2}$ (Figure 3a), according to the relations of the system (1).

$$\begin{cases} \vec{u}_1 = (\cos\varphi_1)\cdot\vec{j}_1 + (\sin\varphi_1)\cdot\vec{k}_1 \\ \vec{u}_2 = -(\sin\psi_2)(\sin\alpha)\cdot\vec{i}_1 - (\sin\psi_2)(\cos\alpha)\cdot\vec{j}_1 + (\cos\psi_2)\cdot\vec{k}_1 \end{cases} \tag{1}$$

From the $\vec{u}_1\cdot\vec{u}_2 = 0$ (perpendicularity of the versors $\vec{u}_1$ and $\vec{u}_2$) condition, the relationship between their projections on the fixed axes (2) is deduced.

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$$-\cos\varphi_1 \cdot \sin\psi_2 \cdot \cos\alpha + \sin\varphi_1 \cdot \cos\psi_2 = 0 \tag{2}$$

Expression (2) is written in the default form (3) in which the rotation angle of the output shaft 2 is based on the angle of rotation of the input shaft 1, but also according to the sharp angle α=δ.

$$tg\,\psi_2 = \frac{1}{\cos\alpha} \cdot tg\,\varphi_1 \tag{3}$$

Then the 0 transmission function gets the expression (4).

$$\psi_2 = arctg\left(\frac{1}{\cos\alpha} \cdot tg\,\varphi_1\right) \tag{4}$$

By derivation the expression of reduced angular velocity (5) is obtained. Initially the expression is also depending on ψ_2, and at the end it is expressed only according to φ_1 (and of course by α).

$$U \equiv \psi_2' = \frac{\dot{\psi}_2}{\dot{\varphi}_1} \equiv \frac{\omega_2}{\omega_1} = \frac{1}{\cos\alpha} \cdot \frac{\cos^2\psi_2}{\cos^2\varphi_1} = \frac{\cos\alpha}{\cos^2\alpha \cdot \cos^2\varphi_1 + \sin^2\varphi_1} \tag{5}$$

By a new derivation is obtained also the order of transmission 2, respectively the reduced angular acceleration (relation 6).

$$W \equiv \psi_2'' = \frac{\ddot{\psi}_2}{\dot{\varphi}_1^2} \equiv \frac{\varepsilon_2}{\omega_1^2} = \frac{-\cos\alpha \cdot \sin^2\alpha \cdot \sin(2\varphi_1)}{\left(\cos^2\alpha \cdot \cos^2\varphi_1 + \sin^2\varphi_1\right)^2} \tag{6}$$

The extreme values of the reduced angular velocity of the driven shaft 2 are obtained from its analytical expression with the limit conditions for the input angle φ_1.

$$\begin{cases} \varphi_1 = 0,\ \ \pi,\ \ 2\pi \Rightarrow \psi_{2\max}' = \dfrac{1}{\cos\alpha} \\ \varphi_1 = \dfrac{\pi}{2},\ \dfrac{3\pi}{2}\ \ \Rightarrow \psi_{2\min}' = \cos\alpha \end{cases} \tag{7}$$

The dual cardan mechanism is obtained by mounting two simple cardan shafts in series, so that the two intermediate shaft forks are coplanar (see Figure 4).

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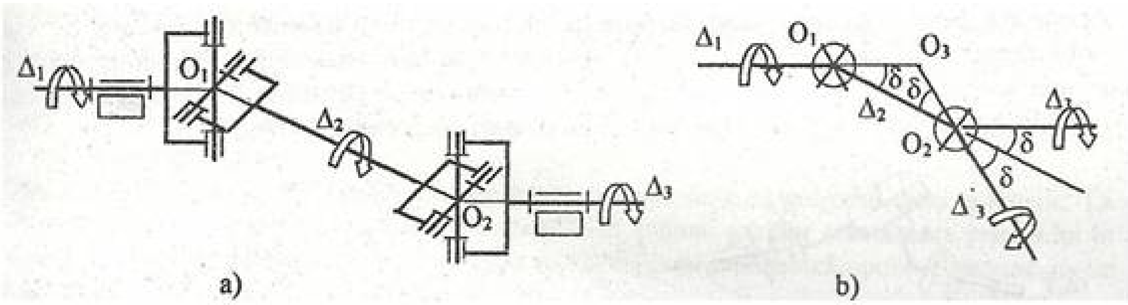

Figure 4: Cardan transmission (double cardan mechanism); the intermediate shaft is observed

3. DOUBLE CARDAN MECHANISM

The double cardan mechanism has the advantage of performing the synchronous movement between the input shafts Δ_1 and the output Δ_3 (Figure 4a). The intermediate shaft with the axle Δ_2 has two fixed points $O_1 \in \Delta_1$ and $O_2 \in \Delta_3$ and so that it no longer needs a material bonded to the fixed bed (Figure 4a).

For the output shaft there are two positions (see Figure 4b): one $\Delta_3 \parallel \Delta_1$, and another $\Delta_3^{'}$ symmetrical with the axis Δ_3 in relation to the axis Δ_2. The synchronization of the two movements (input-output) can be proved by means of the 0 transmission function which is written for the two simple cardan couplings (figure 4a), (relation 8); (Petrescu, 2012b).

$$tg\psi_2 = \frac{1}{\cos\alpha} \cdot tg\varphi_1 = \frac{1}{\cos\alpha} \cdot tg\psi_3 \Rightarrow tg\varphi_1 = tg\psi_3 \Rightarrow \psi_3 = \varphi_1 \quad (8)$$

The dual cardan coupling variant with the parallel axle output and input is generally used on heavy goods vehicles (lorries) so that the distance between the two axes may vary within certain limits (see Figure 5); In this situation a variable length of the intermediate shaft 2 is required, constructively made by a telescopic intermediate shaft.

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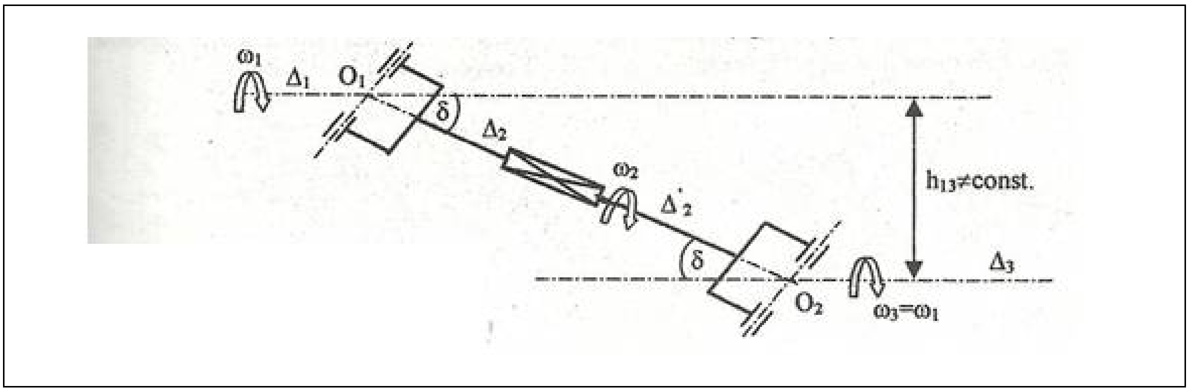

Figure 5: Double, synchronous and telescopic (with telescopic intermediate shaft)

The two pieces of the axle 2, Δ_2 şi Δ_2', have the coplanar forks (Figure 5) and are connected by a transverse groove coupling, allowing the variation of the length O_10_2 and the angle dimension $\alpha=\delta$ when the distance between the axes h_{13} varies.

If the two forks of the intermediate shaft Δ_2 are not coplanar (see figure 6), the movement of the driven shaft Δ_3 is no longer synchronous with that of the drive shaft Δ_1.

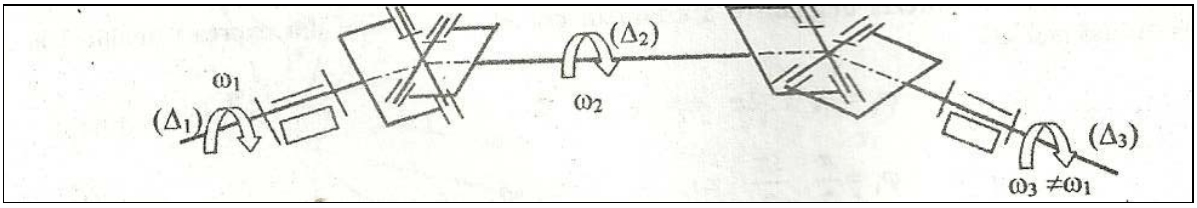

Figure 6: Double Asynchronous Cardan Transmission (intermediate shaft forks are not coplanar)

4. CONCLUSIONS

Although apparently the dynamic loading of the double (dual) transmission increases with the mechanical (mechanical) inertial moment, the effect itself is negligible under actual operating conditions (for normal cardan transmissions, built and properly mounted).

The elasticity of the intermediate shaft influences the homokinetic deviation of the transmission as follows: a) in the usual cases the bicardan transmission becomes quasi-homo-kinetic and therefore the deviation from homokinetic can be virtually neglected; b) In special cases with long (or very long) intermediate shafts and high (or very high) mechanical moments of inertia it is necessary to offset the homokinetic deviation by designing the transmission so that the deviation from homokinetic becomes null, or negligible. c) Under normal operating conditions, the

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influence of the elasticity of the intermediate shaft on torsion moments may be neglected.

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7. AUTHORS' CONTRIBUTION

All the authors have contributed equally to carry out this work.

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